1992

A Simple Valve Model to Study the Performance of a Small Compressor

K. T. Ooi
Nanyang Technological University

G. B. Chai
Nanyang Technological University

E. C. Kwek
Matsushita Refrigeration Industries; The Republic of Singapore

A SIMPLE VALVE MODEL TO STUDY THE PERFORMANCE OF A SMALL COMPRESSOR

K.T., OOI and G.B. Chai
Nanyang Technological University
Singapore 2263

E.C., KWEK
Matsushita Refrigeration Industries (S) Pte Ltd
1 Bedok South Road
Singapore 2263

ABSTRACT

Reed valves in many small compressors are usually activated automatically by pressure differences across the valve. These valves are sometime called automatic valves. They have a very simple geometry and construction but they comprise one of the most complicated performance characteristics. The unsteady vibration of the valve reed during the operation requires a proper design. Generally too stiff the valve plate causes over-compression and too flexible causes unnecessary fluctuation of the valve plate.

The paper presents a simulation model of a typically used reed valve in a small refrigeration rolling piston compressor. The natural fundamental frequency was obtained using the Rayleigh's quotient. The non-uniformity of the valve width can be conveniently accounted for by considering the deflected mode shapes of the valve reed using the actual valve geometry and dimensions. The mode shape can then be applied in the simulation model of the valve reed. The mode shapes for the deflected valve before and after hitting the valve stop are taken into the consideration. Predicted results are presented for various compressor delivery pressures. Effects of various valve damping coefficient, valve thickness and sensitivity of the valve behaviour with respect to the valve force area coefficient are also presented.

NOMENCLATURE

A_v	Valve flow area, m^2
B	effective valve force area, m^2
C_d	Coefficient of discharge
C_{fa}	force area coefficient
d	valve body width, m
E	Young modulus, N/m^2
I	moment of inertia, m^4
h_1	upstream enthalpy, kJ/kg
h_2^{u}	downstream enthalpy assuming isentropic expansion, kJ/kg
l	effective valve length, m
$\dot{m}$	mass flowrate, kg/s
P	pressure, N/m^2
P_{ind}	indicated power, W
P_{over}	over compression power loss, W
$q(x)$	mode participation factor
r	valve tip radius, m
t	valve thickness, m
v_s	specific volume, m^3/kg
$\overline{V}$	flow velocity, m/s
x	distance from valve free end, m
$y(x)$	deflection of valve
ω	natural frequency
$\phi(x)$	valve shape function
ρ_p	plate density, kg/m^3
ζ	damping coefficient

1. INTRODUCTION

Compressor valves have received much attention over the improvement study and the reliability of the compressor. Comprehensive study on various aspects of the valve behaviours may be found in literatures [4-10]. Most of these paper studied the vibration response, dynamic stress and noise related problems. This paper attempts a simple simulation of a single port reed valve configuration to assess the effects of its design parameters and operating coefficients on the performance of the compressor.

Fig. 1 shows a reed valve sitting on a discharge port of a rolling piston compressor. In this compressor, because of the nature of its operation, the existence of the discharge valve is necessary to prevent backflow at the beginning of the compression process. The opening and the closing of the valve are caused by interactions between fluid pressures as well as valve dynamic characteristics. The design of this component generally requires information on valve flow characteristics as well as stress and the fatigue behaviour of the valve material.

In the simulation of the compressors as a whole, the need to simulate the valve flow characteristics is to determine the discharge port opening position and the discharge characteristic of the compressor. The valve dynamic characteristic will affect the performance of the compressor.

If the flow through the valve is assumed isentropic, the flow velocity may be calculated:

$$\overline{V} = \sqrt{2(h_1 - h_2^{s})} \qquad \ldots(1.1)$$

The mass flowrate can be found by,

$$\dot{m} = \frac{C_d A_v \overline{V}}{v_s} \qquad \ldots(1.2)$$

Where C_d is the discharge coefficient which accounts for the effective flow area as well as the isentropic efficiency for an actual non-isentropic flow. The choke flow condition may be taken into consideration by taking the maximum flow velocity to be equal to the sonic velocity at the throat of the port. The effective flow area $C_d A_v$ may be calculated from the valve opening annular area or valve port area. The smaller flow area imposes higher restriction to the flow hence it should be taken as an effective flow area in the valve flow calculation.

2. VALVE REED MODEL

2.1 Introduction

Fig. 2 shows the geometry of the valve plate. Its geometry may be simplified by assuming the valve free end as a circular disc and is connected to a rectangular valve plate. The valve displacement is constrained by the valve backing plate which is placed a small distance above the valve plate. The function of this backing plate is to prevent the valve plate from over stressed, which may be due to unnecessary large deflection.

2.2 Valve Dynamics

The dynamic of the valve is characterised by the vibration of the valve plate. Assuming the valve as a beam with varying width, neglecting shear and rotary motion, the beam equation describing the beam vibration [3] is:

$$\frac{\partial^2}{\partial x^2}\left(EI(x)\frac{\partial^2 y}{\partial x^2} + \rho_p A(x)\frac{\partial^2 y}{\partial t^2} \right) = P(x,t) \qquad \ldots(2.1)$$

And assuming the valve deflection is given by:

$$y = \sum_{n=1}^{\infty} \phi_n(x) q_n(t) \qquad \ldots(2.2)$$

Where $\phi_n(x)$: valve shape function which may be determined from the free vibration analysis. There exists infinite number of combinations of mode shapes since n ranges from 1 to infinity, but in many cases especially for the valve flow area calculation, the first mode consideration alone may be sufficient [1].

$q_n(t)$: generalised coordinate or mode participation factor which may be obtained by integrating the valve vibration governing equation using an initial value integration technique.

The mode participation factor is obtained from:

$$\ddot{q}_n + 2\zeta\omega_n\dot{q}_n + \omega_n^2 q_n = \frac{\int \phi(x)_n P\,dx}{\rho_p A\,\phi(x)_n^2 dx} \quad \ldots(2.3)$$

Where ζ_n is the damping coefficient of the valve reed due to the air cushioning.

For a single valve port configuration and to avoid considerations of the flow in the valve gap, the gas force acting on the valve may be taken to be equal to the product of the effective valve force area and the pressure dropped across the valve [1],

$$\int \phi_n(x) P\,dx = \phi_n(x) B(x) \Delta P(t) \quad (2.4)$$

Where B(x) is the effective force area and the $\Delta P(t)$ is the pressure difference across the valve. Generally the term $B(x)$ has to be obtained from experiments. It is a function of valve lifts.

Substituting eqns. (2.4) into (2.3), the following expression may be obtained:

$$\ddot{q}_n + 2\zeta\omega_n\dot{q}_n + \omega_n^2 q_n = \frac{\phi_n(x) B(x) \Delta P(t)}{\rho_p A\,\phi(x)_n^2 dx} \quad \ldots(2.5)$$

In the above equation the natural frequency of the valve may be obtained approximately from the Rayleigh's quotient. i.e.

$$\omega_n^2 = \frac{X}{Y} \quad \ldots(2.6)$$

where

$$X = E\int_0^{x_1} I(x)\left(\frac{d^2\phi(x)}{dx^2}\right)^2 dx + EI\int_{x_1}^{l}\left(\frac{d^2\phi(x)}{dx^2}\right)^2 dx \quad \ldots(2.7)$$

$$Y = \rho_p \int_0^{x_1} A(x)\phi(x)^2 dx + \rho_p A\int_{x_1}^{l} \phi(x)^2 dx \quad \ldots(2.8)$$

where the values of $A(x)$ and $I(x)$ are given below,

$0 < x < x_1$	$x_1 < x < l$
$A(x) = 2t\sqrt{2rx - x^2}$	$A = dt$
$I(x) = \frac{t^3}{6}\sqrt{2rx - x^2}$	$I = \frac{dt^3}{12}$

For a better approximation, the deflected mode shape may be obtained from a finite element model employing the actual geometry of the valve reed. In the above solution, the valve lift are monitored to determine if the valve hits the backing plate and correct boundary conditions are imposed. Two cases were considered: one is when the valve leaves the seat, the other is when the valve hits the valve backing plate [2].

In the present model only the first mode of valve deflection is considered, the model however takes into considerations different boundary conditions as well as valve mode shapes when the valve departs from the valve seat, and when it hits and departs from the valve backing plate. Fig. 3 shows the two different valve mode shapes.

In the present analysis, the shape function for the case when the valve is departed from the valve seat is obtained from a standard polynomial function for a cantilever beam [3]:

$$\phi(x)=\left(\frac{x}{l}\right)^4-4\left(\frac{x}{l}\right)+3 \qquad ...(2.9)$$

In eqn. (2.9) the natural and geometric boundary conditions for both ends are satisfied:

$$\text{at} \quad x=0, \quad \frac{d^2\phi(0)}{dx^2}=0, \frac{d^3\phi(0)}{dx^3}=0, \quad \text{and} \quad \text{at} \quad x=l, \quad \phi(l)=0, \quad \frac{d\phi(l)}{dx}=0$$

When the valve hits the backing plate, the following standard polynomial function is applied [3]:

$$\phi(x)=\left(\frac{x}{l}\right)^4-\frac{3}{2}\left(\frac{x}{l}\right)^3+\frac{1}{2}\left(\frac{x}{l}\right) \qquad ...(2.10)$$

Notice that in eqn. (2.10) only the geometry boundary conditions are satisfied. It satisfies the boundary conditions: at $x=0$, $\phi(0)=0$, $d^2\phi(0)/dx^2=0$ and at $x=l$, $\phi(l)=0$, but it does not satisfy $d\phi/dx=0$.

3. RESULTS AND DISCUSSIONS

Before the valve model is coupled to the compressor model to predict the valve flow characteristics, its accuracy is checked against a finite element simulation results. This is because the present analysis employs only the fundamental mode shape and the mode shape given by eqn. (2.10) does not satisfy all the natural boundary conditions. The finite element analysis employs small elements over the entire valve plate of the actual valve geometry and capable of predicting various deflected modes. It is also believed that the solution given by the finite element analysis is more accurate as it accounts more closely to the actual plate characteristics. The comparison may indicate the order of accuracy of the simulation results between a simplified model and a more comprehensive finite element model.

Good agreements are obtained from the two models on the values of the valve fundamental mode of frequency. The deflection results show that there exists small discrepancies between the two models, see Fig. 4. The simplified model predicts higher deflections near the clamped end. These discrepancies may be due to the boundary condition which is not satisfied in applying the shape function, as mentioned before. Such discrepancies may be acceptable in the present approximation for valve flow area determination in view of the estimations that have to be made for the valve force area coefficient and the damping coefficient once the valve is coupled with the compressor model. A more accurate estimation of deflection may be required to perform a stress analysis on the reed valve.

In this section the effects of valve thickness, damping coefficient and force coefficient on the valve characteristic and their effects on the compressor performance will be discussed. In general, it is commonly known that a highly oscillatory valve characteristic is unwanted because it shortens the valve working life (due to fatigue) as well as causes higher noise level. On the other hand, too stiff the valve reed caused by a thicker valve, may result in too small valve displacements (see Fig. 5, the displacements shown are at the valve port center, not at the valve tip) which in turn may cause a higher over compression loss (see Figs. 6 and 8) and this in turn results in a higher discharge velocity at the end of the discharge process. The latter effects is shown by Fig. 7 with a rapid mass reduction at the end of discharge process indicating a higher pressure differential across the valve (see Fig.6). In practice this higher flow velocity may cause a higher turbulence intensity and hence resulted in a higher noise level [5].

Fig. 9 shows the effects of valve thickness on the over compression loss as well as the total indicated power. The results shown are normalised to that for valve with 0.2mm thickness. The results show that the values of indicated power P_{ind} and the over compression loss P_{over} increase with the valve thickness. Judging from the pressure variation in Fig. 6 and chamber mass variation in Fig. 7, the valve with thickness of 0.2mm appears to be the optimum.

Results also revealed that a thinner valve reed, causes no restriction to discharge flow and hence has a minimum over compression loss. However in practice, too thin a valve reed causes unnecessary high valve displacements which shortens the reliability of the valve working life and it may not provide a proper sealing of the valve port during the suction and the compression stages.

In practice, the damping effects on the valve plate may be caused by viscous effects or fluid cushioning effects. In a simple simulation, the value of valve damping coefficient may be chosen so that the predicted valve characteristic agrees well with the measured results. In this preliminary analysis, attempts have been made to assess the sensitivity of the damping coefficient on the valve characteristic and hence the compressor performance. The results on the valve characteristic and compressor performance for various valve damping coefficients are presented, see Figs. 10-11. Generally results show that small valve damping coefficients (say 0.1 and below) causes unnecessary oscillatory motions while higher damping coefficients resulted in a smoother valve operation, as would be expected. Results also show that these effects may affect the over compression loss. However the effects is insignificant, see Fig. 12.

In the compressor valve simulation, usually the value of the force area coefficient are also measured experimentally. In the present study, the results for various values of this force area coefficient are presented to demonstrate the effects of this parameter on the valve characteristic as well as the performance of the compressor, see Figs. 13-14. Results showed that smaller value of force area coefficients cause higher magnitude of valve oscillations and vice-versa. These effects resulted in fluctuations in the compressor chamber pressure, during the discharge process. These effects also affect the over compression loss, but the effects on the overall indicated power is negligible, see Fig. 15.

4. CONCLUSION

A simple simulation model for the reed valve may be used in the compressor simulation for performance predictions. To assess the accuracy of the model, the predictions should be compared against experimentally measured results. In the compressor performance prediction, the values of valve damping coefficient and effective valve force area may be estimated from the experiment.

The simulation model shows that valve thickness has marked effects on the compressor performance. There exists an optimum thickness which compromises between the optimum valve flow area and the reliability of the valve reed. The prediction also shows that the valve damping coefficient as well as valve force area coefficient may effect the over compression loss, however the effects are insignificant over the range examined, as compared to that of the valve thickness.

5. ACKNOWLEDGEMENT

The authors wish to thank Matsushita Refrigeration Industries (S) Pte Ltd for the supports provided. Thanks are due to Mr Noriaki Okubo and Mr Khoo Chew Thong for providing valuable time and help during the study.

6. REFERENCES

1. W., Soedel, Introduction To Computer Simulation Of Positive Displacement Type Compressors, Course Note, Purdue University
2. J.F., Hamilton, Extension of Mathematical modelling of positive displacement type compressors, Course Note, Purdue University
3. W.T. Thompson, Theory of Vibration with Applications, George Allen & Unwin, 1983
4. D.S., Schwerzler, J.F. Hamilton, An Analytical Method For Determining Effective Flow And force Areas For Refrigeration Compressor Valving Systems, Proceedings of the 1972 Purdue Compressor Technology Conference, 1972
5. T.J., Trella, W., Soedel, On noise Generation of Air Compressor Automatic Reed Valves, Proceedings of the 1972 Purdue Compressor Technology Conference, 1972
6. R., Cohen, Proceedings of the 1972 Purdue Compressor Technology Conference, 1972
7. A. M., Seligman, Simplified theory Applied to automatic Compressor Valves, Proceedings of the 1972 Purdue Compressor Technology Conference, 1972

8. S., Paspastergiou, J. Brown, J.F.T., Maclaren, Impact Velocities of Valve Reeds, Proceedings of the 1982 Purdue Compressor Technology Conference, 1982
9. L., Boswirth, Theoretical And Experimental Study on Valve Flutter, Proceedings of The 1990 International Compressor Engineering Conference At Purdue, 1990
10. J., Piechna, A Computational Study Of The Interaction Between Fluid Flow And Reed Valve Deformation, Proceedings of the 1990 International compressor Engineering Conference at Purdue, 1990

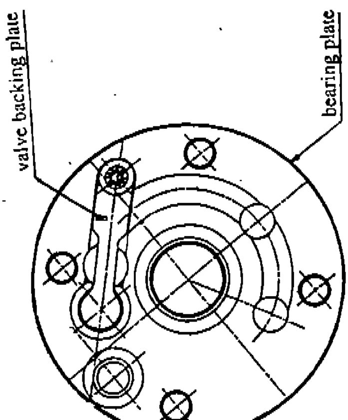

FIG. 1 COMPRESSOR AND VALVE REED

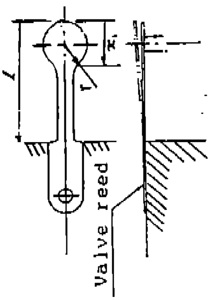

FIG. 2 GEOMETRY OF THE VALVE REED

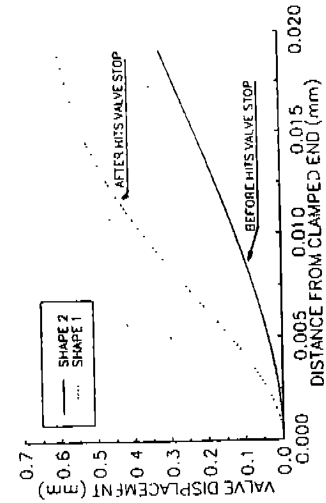

FIG. 3 DIFFERENT VALVE SHAPE FUNCTIONS

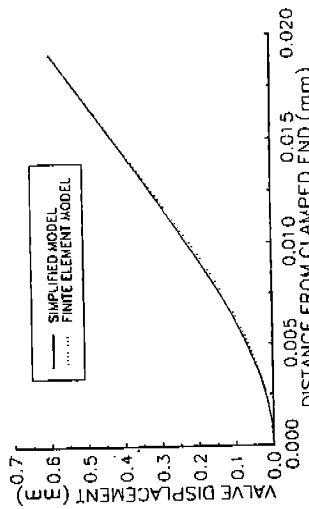

FIG. 4 COMPARISON OF VALVE DISPLACEMENTS

FIG. 5 VALVE DISPL. DURING DISCHARGE PROCESS
(AT PORT CENTER)

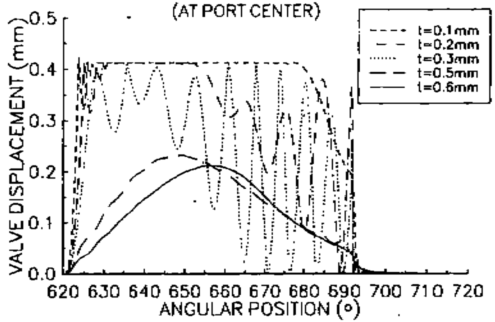

FIG. 6 CHAMBER PRESS. DURING DISCHARGE PROCESS

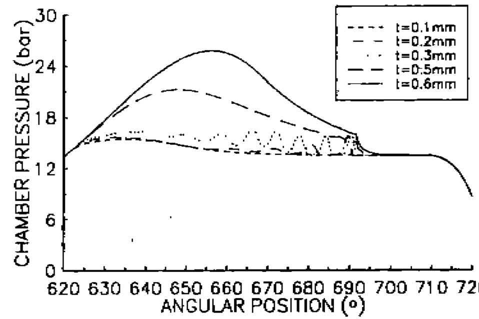

FIG. 7 CHAMBER MASS DURING DISCHARGE PROCESS

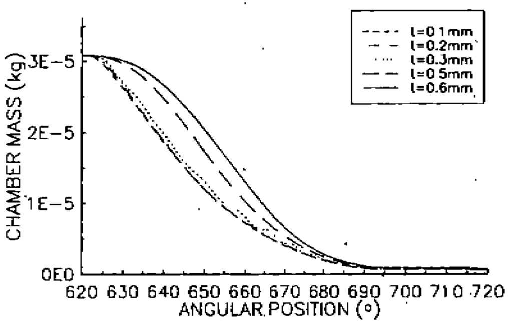

FIG. 8 PRESSURE-VOLUME DIAGRAMS

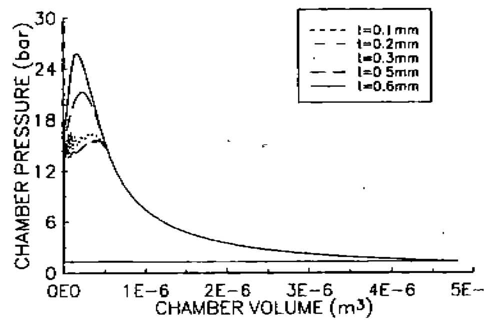

FIG. 9 OVER–COMPRESSION LOSS & INDICATED POWER
(RESULTS NORMALISED TO t=0.2mm)

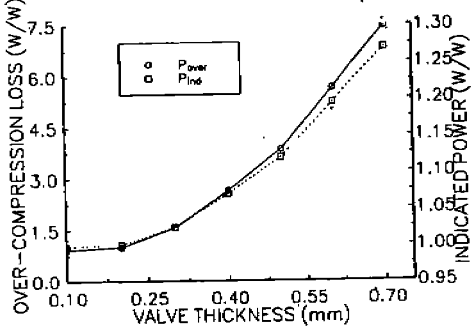

FIG. 10 VALVE DISPL. DURING DISCHARGE PROCESS
(AT PORT CENTER)

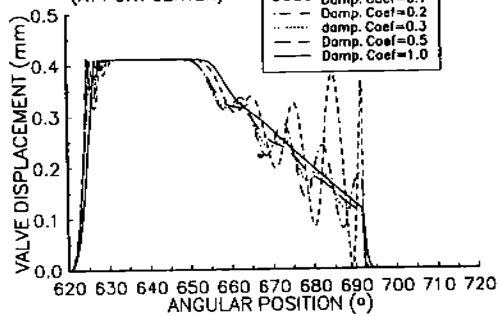

FIG. 11 CHAMBER PRESS. DURING DISCHARGE PROCESS

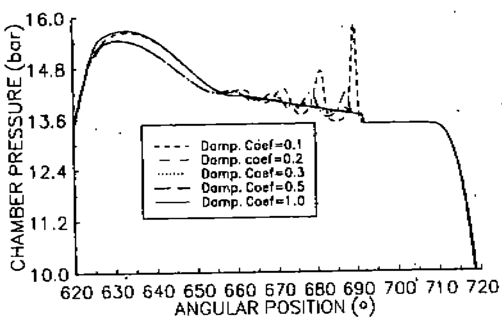

FIG. 12 OVER–COMPRESSION LOSS & INDICATED POWER
(RESULTS NORMALISED TO Damp. Coef=0.1)

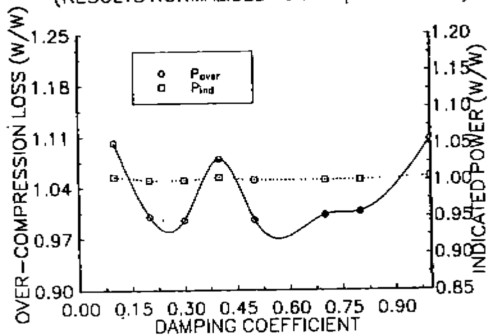

FIG. 13 VALVE DISPL. DURING DISCHARGE PROCESS
(AT PORT CENTER)

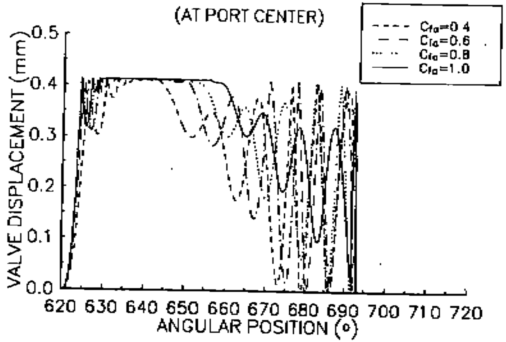

FIG. 14 CHAMBER PRESS. DURING DISCHARGE PROCESS

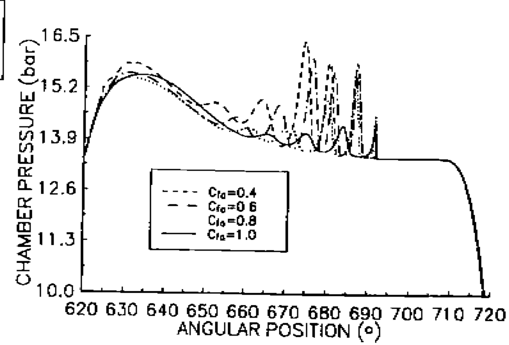

FIG. 15 OVER–COMPRESSION LOSS & INDICATED POWER
(RESULTS NORMALISED TO C_{fa}=0.7)